Review

Design Improvement of a Viscous-Spring Damper for Controlling Torsional Vibration in a Propulsion Shafting System with an Engine Acceleration Problem

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Received: 12 May 2020; Accepted: 3 June 2020; Published: 11 June 2020

Abstract: In order to cope with strengthened marine environmental regulations and to reduce fuel consumption, recently constructed vessels are equipped with an ultra-long stroke engine and apply engine de-rating technology. This was intended to improve propulsion efficiency by adopting a larger diameter propeller turning at a lower speed but also results in a significant increase in the torsional exciting force. Therefore, it is very difficult to control the torsional vibration of its shaft system by adopting a damper, for ships equipped with fuel-efficient ultra-long-stroke engines, even though previously, torsional vibration could be controlled adequately by applying tuning and turning wheels on the engine. In this paper, the vibration characteristics of an ultra-long-stroke engine using the de-rating technology are reviewed and dynamic characteristics of a viscous-spring damper used to control the torsional vibration of its shaft system are also examined. In case of ships have recently experienced an engine acceleration problem in the critical zone, it is proposed that the proper measures for controlling torsional vibration in the propulsion shafting system should include adjusting the design parameters of its damper instead of using the optimum damper designed from theory in order to prevent fatigue fracture of shafts.

Keywords: ultra-long stroke engine; engine acceleration problem; torsional vibration; propulsion shafting system

1. Introduction

Due to the impact of the International Maritime Organization (IMO) enforcement measures, recently built vessels are equipped with ultra-long-stroke engines [1–3]. This engine has been developed to improve propeller efficiency and utilize engine de-rating technology to improve the thermodynamic efficiency of the engine [4]. This type of engine can generate greater torque compared to the previous super-long stroke engine, yet it is known to increase the torsional exciting force by up to 40%. Therefore, it is barely possible to control the torsional vibration of shaft system by adopting a torsional vibration damper, for ships equipped with the fuel efficient ultra-long-stroke engine, even though previously, the torsional vibration could be controlled adequately by applying tuning and turning wheels on the engine [5–8]. In addition, when the de-rating ratio applied to improve fuel efficiency is excessive, the torque limit line is lowered parallel to that of a standard engine, causing a decrease in the engine acceleration performance in the low load range. This phenomenon affects ship stability, so it is fatal to a propulsion shafting system in which the barred speed range (hereafter BSR) is set within the operating speed range. That is, a decrease in the acceleration capability of the engine causes the ship to stay in

the BSR longer than necessary. In such cases, there is a possibility of fatigue fracture, such as that resulting from a reduction of shaft fatigue life [9–11]. In general, a torsional vibration damper is not designed to control this aspect of the torsional vibration problem [5–8]. Thus, for shaft systems that might be exposed to acceleration delay while in the BSR, proper measures should be taken to avoid fatigue fracture during the shaft life expectancy.

Han et al. performed an effective experimental evaluation of shaft fatigue life for torsional vibration in the input shaft of a reduction gear. The gear was connected to a diesel engine with the Soderberg criteria and subject to the linear damage accumulation law. Data was collected by measuring directly with a strain gauge and a telemetry system. They reported that the variations in the torsional vibration, revolution speed of the shaft, and acceleration of the diesel engine could be increased when the backpressure in the exhaust pipe of the diesel engine was dramatically increased. This condition depends on the sailing condition such as zig-zag and straight operating modes [12]. Barro et al. reviewed and evaluated the fatigue life of the intermediate shaft using theoretical analysis and measurement data of the torsional vibration for the propulsion shafting system of a ship equipped with a 6G70ME engine designed and manufactured by MAN Energy Solution [7]. Kim et al. investigated the dynamic characteristics of the shafting system in ships with an engine acceleration problem in the barred speed range and then examined the fatigue life of the corresponding shafts using a detailed evaluation method based on stress-life (hereafter, S-N) methodology. They proposed an alternative design method to enable safer ship operation by an appropriate change in the design of the corresponding shafts considering the torsional vibration characteristics of each [9]. Det Norske Veritas and Germanischer Lloyd (hereafter DNVGL) has developed guidelines for evaluating the fatigue life of power transmission shafts (using a detailed evaluation method based on S-N methodology) if the BSR cannot be passed within five seconds [13]. The engine manufacturer (MAN Energy Solution) has developed a dynamic limiter function (hereafter, DLF) technology that enhances engine acceleration capability in the low-load range where the BSR is and provides DLF as a standard provision for ships equipped with fuel efficient ultra-long-stroke engines. However, it also causes an increase in the thermal load and the vibratory torque [14,15].

In this paper, the theory governing the dynamic characteristics and performance of a viscous-spring damper used to control the torsional vibration of a propulsion shafting system is reviewed and then examined in detail to determine what vibration characteristics might be used to optimize the viscous-spring damper. In addition, considering the vibration characteristics of the propulsion shafting system for ships with the possibility of acceleration delay while in the BSR, it is proposed that the proper measures for controlling the torsional vibration in the propulsion shafting system should include moderate adjustments of the design parameters of its damper instead of using a damper design based on optimum damper design theory.

2. Characteristics of the Torsional Vibration of the Propulsion Shafting System with rpm Acceleration Problem

2.1. Characteristics of Torsional Exciting Torque in an Ultra-Long-Stroke Engine with De-Rating Technology

The fuel efficient ultra-long-stroke engine that was developed to improve propulsion efficiency by matching low-speed with a propeller of large diameter, can generate great power even at low speed, yet has the disadvantage of inducing larger torsional exciting force [3,9]. Figure 1 shows the comparison result of the torsional vibratory torque of two engines with the same cylinder diameter but different strokes. Here, Case I represents an engine with a stroke of 2800 mm, and Case II represents an engine with a stroke of 3256 mm. Looking at this figure, it can be seen that the torsional exciting torque increases by up to 40% as the stroke increased by 16% compared to past stroke lengths. In addition, the engine de-rating technology used to optimize its engine power under specific operating conditions by deliberately reducing the specific maximum continuous rating (SMCR) was applied to its engine with a combination of tuning technologies to optimize the combustion characteristics under partial loads [3–9].

Figure 1. Comparison of sixth-order vibratory torque between two engine models with different stroke values.

Figure 2a shows the de-rating ratio of the engine installed on an oil tanker and bulk carrier that are currently being built. It can be seen from this figure that the engine power was decreased by 20% to 40%. However, this decrease was previously limited to a maximum of 10% until fuel efficient ultra-long-stroke engines were developed. These changes were intended to increase the thermal efficiency and to reduce fuel consumption by making the maximum engine power and mean effective pressure lower than that of standard engines while keeping the combustion pressure at a maximum. Therefore the lower the engine rated power, the lower the mean effective pressure. Reduction of the mean effective pressure to improve fuel efficiency has become a key factor in the increase of the torsional exciting torque at the same mean effective pressure, as shown in Figure 2b. In conclusion, the lower the engine rated power, the greater the torsional exciting torque at the same mean effective pressure.

2.2. Dynamic Characteristics of Transient Torsional Vibration of the Propulsion Shafting System with Acceleration Delay while in the Barred Speed Range

Figure 3a shows a schematic diagram of the equipment used for measuring the transient torsional vibration of the propulsion shafting system. In order to measure the torsional vibration of the propulsion shafting system, an encoder connected to the forward side of the crankshaft was used and the velocity signal was transmitted from the intermediate box connected to this encoder. This signal was modulated through the F/V converter and analyzed using by signal analyzer (OR-36) manufactured by OROS. Figure 3b,c shows the passing time and transient torsional stress in the BSR. Here, Case III represents a scenario in which the engine speed is increased, and Case IV represents a scenario in which the engine speed is reduced. From this figure, the time required to pass through this BSR is about 73 s when the engine speed is increased and 16 s when the engine speed is reduced. It can also be seen that the maximum torsional stress for the intermediate shaft measured in this BSR is 92.1 N/mm^2 at the resonance point (33.3 rpm) when the engine speed is increased. In general, this range is set as a precondition for rapid passing, and DNG-GL reported that the BSR must be passed within five seconds (DNV-GL, 2015). This ship is equipped with an ultra-long-stroke engine to improve propulsion efficiency, and the engine rating ratio is 24.0% lower than that of a standard engine. Thus, the allowable power for acceleration in the low load range is decreased as the torque limit line is reduced in parallel.

Therefore, as shown in Figure 3b, it took more than one minute to pass the BSR. For cases when this phenomenon does occur, there must be concern about the fatigue fracture of shafts caused by the accumulation of excessive fatigue during their expected lifetime.

Figure 2. (a) De-rating ratio of ships with an ultra-long-stroke engine. (b) Comparison of sixth-order vibratory torque of an ultra-long-stroke engine with de-rating technology according to varied mean effective indicated pressure.

Figure 3. (a) Schematic diagram for torsional vibration measurement. (b) Measured result of duration time within barred speed range. (c) Measured result of torsional vibration in the intermediate shaft within the barred speed range equal to the transient state.

3. Theoretical Analysis Method for the Viscous-Spring Damper Design

For a corresponding engine and shaft system, a multiple-degrees-of-freedom system was replaced with an equivalent one-degree-of-freedom system using the HOLZER method as shown in Figure 4a,b [16–19]. This was done to obtain the natural angular frequency and relative amplitude when the torsional vibration of a corresponding shaft is controlled by adopting the viscous-spring

damper. Once this damper is applied to an engine with the one-degree-of-freedom system, it becomes equivalent to a two-lumped mass system, as shown in Figure 4b [16–19]. The dynamic characteristics of the viscous-spring damper can be described as follows by replacing the viscous-spring damper and shaft system with an equivalent two-lumped mass system. The following is the equation of motion for the vibration system.

Figure 4. (a) Multi degree modeling of the propulsion shafting system in a ship (b) Equivalent mass-spring system with viscous-spring damper. (c) Effect of a viscous-spring damper on the response of a vibratory system.

Generally, the damping coefficient (μ_E) could be neglected in a high-speed combustion engine because the damping coefficient is about 0.03 without a damper. Moreover, it becomes even smaller if the amplitude is reduced when the damper is installed. Thus, the damping coefficient does not have to be considered unless it is a special case. The dynamic magnitude can be expressed as in Equation (3), when $\mu_E = 0$.

Figure 4b shows the relation between $M_R (= \theta_E / \theta_{st})$ and the frequency ratio γ in accordance with the varied damping coefficient ratio μ , where $\lambda = 1.0$ and $R = 1/20$. As shown in Figure 4c, the purpose for the application of the damper is to reduce the amplitude peak as much as possible when resonance exists. To accomplish this, the damping coefficient should neither be too small nor too large. Thus, the optimum damping coefficient, one that achieves the lowest resonance amplitude peak, has to be found when the vibration energy dissipated by heat becomes maximum. The natural frequency ratio μ that allows the points P and Q to have even height (θ_E / θ_{st}) can be obtained using Equation (4) because all corresponding resonance curves correlated to each damping coefficient ratio μ , pass through points P and Q. Equation (5) is the dynamic magnitude when points P and Q have even height. Here, the optimum damping coefficient μ (maximizes the height of P and Q) is derived in Equation (6).

$$J_E \ddot{\theta}_E + K_E \theta_E + K_d (\theta_E - \theta_d) + C_E \dot{\theta}_E + C_d \dot{\theta}_d = P_o \cos \omega t \quad (1)$$

$$J_d \ddot{\theta}_d + K_d (\theta_d - \theta_E) + C_d (\dot{\theta}_d - \dot{\theta}_E) = 0 \quad (2)$$

$$M = \sqrt{\frac{(2\mu\gamma)^2 + (\gamma^2 - \lambda^2)^2}{\mu^2 [2\gamma(\gamma^2 - 1 + R\gamma^2)]^2 + [R\lambda^2\gamma^2 - (\gamma^2 - 1)(\gamma^2 - \lambda^2)]^2}} \quad (3)$$

$$\lambda = \frac{1}{1 + R} \quad (4)$$

$$M = \left(\frac{\theta_E}{\theta_{st}} \right)^2 = 1 + \frac{2}{\mu} \quad (5)$$

$$\mu = \sqrt{\frac{3R}{8(1 + R)^3}}. \quad (6)$$

4. Analysis of Results and Discussion

In this section, the characteristics of the torsional vibration are analyzed and a control method are examined for a propulsion shafting system equipped with an ultra-long-stroke engine developed to improve its fuel consumption in the high oil price era. This ship took several minutes to pass the BSR in a range of operating speeds because of inadequate torque in the low-load range. This resulted from excessive de-rating. In such cases, there is a possibility of fatigue fracture, such as that resulting from reduction of the shaft fatigue life. In general, the torsional vibration damper is not designed to control this aspect of the torsional vibration problem. Thus, for shaft systems that might be exposed to acceleration delay in the BSR, proper measures should be taken to avoid fatigue fracture during its expected lifetime (20–25 years). Here, some cases of the acceleration delay are examined to inform the optimum design of the torsional vibration damper

4.1. Dynamic Characteristics of an Existing Viscous-Spring Damper and Torsional Vibration for a Propulsion Shafting System

Table 1 shows the specifications of the corresponding shaft system and Table 2 shows the specifications of the torsional tuning damper installed in front of the main engine. In the case of this corresponding shafting system, the node point of the first mode shape exists in the intermediate shaft, and the torsional stress of its shaft exceeds the limit for transient operation. This is equal to the yield strength of the corresponding shaft, as recommended by the classification societies. Therefore,

the damper is applied to control the torsional vibration of the shaft system to reduce the torsional stress of the intermediate shaft. Figure 5a shows the dynamic characteristics of the existing damper applied to the vessel in the case study. As shown in Figure 5a, the dynamic magnitude is as high as 14.3. Thus, after calculating the optimum stiffness and damping coefficient ratio by the process of optimum damper design theory, and maintaining the current inertia value of the damper, the results are 7.156 MNm/rad and 0.141, respectively. This outcome is 71.6% and 49.5% lower than those of the existing design. Figure 5b shows the dynamic characteristics of the modified damper applying the optimum stiffness and damping coefficient, and calculated using optimum damper design theory. As shown in Figure 5b, the dynamic magnitude is about 5.7, which is 60.0% lower than that of the existing damper. In this research, the existing damper design was modified to include an optimum stiffness and damping coefficient ratio based on optimum damper design theory.

Table 1. Specification for the propulsion shafting system.

	Item	Detail
Main engine	Type	6G70ME-C
	MCR (kW × rpm)	16,590 × 77.1
	Cylinder bore (mm)	700
	Stroke (mm)	3256
	MEP (bar)	18.4
	Ratio of connecting rod	0.5
	Reciprocating mass (N/cylinder)	94,987
	Turning wheel (kg·m ²)	15,000
Shaft	Intermediate shaft (mm)	525/0
	Propeller shaft (mm)	670/0
Propeller	Type	FPP
	Number of blades	4
	Diameter (m)	8.8
	M.O.I. in water (kgm ²)	180,090

Table 2. Specifications for torsional vibration damper.

Item	Detail
Type	D330/EU
M.O.I. of damper outer ring (kgm ²)	16,800
inner ring (kgm ²)	1.340
Stiffness (MNm)	10.0
Relative damping coefficient (Nms/rad)	210,000
Permissible elastic torque at Continuous condition (kNm)	439
Permissible elastic torque at transient condition (kNm)	659
Permissible thermal load (kW)	230
Oil flow (L/min)	170
Weight (N)	214,766

Figure 6a,b shows the vibration characteristics of its shaft system with the modified damper design. Here τ_1 is the limit line for continuous operation and τ_2 is the limit line for transient operation recommended by the IACS (International Association of Classification Societies) and global shipping registries such as American Bureau of Shipping(hereafter ABS), DNVGL, Lloyd's Register(hereafter LR), and Korean Register(hereafter KR) [20–24]. The torsional stress in the intermediate shaft can be reduced to 90.1 N/mm², which is 2.0% lower than that of the existing damper design and the resonance point increases by 2 rpm to 36 rpm. In addition, the torsional stress in the propeller shaft can be reduced to 43.2 N/mm² which is 2.0% lower than that of the existing damper design and the resonance point increases by 2 rpm to 36 rpm, yet the prohibited speed range increases by 1 rpm. It is not only

that the optimum damper theory is applied to reduce the torsional stress on the shafts, but also that the range of the rpm that exceeds the limit for continuous operation is rather increased. Usually, it is suitable to apply optimum damper design theory to reduce the torsional stress of the intermediate shaft. However, in the case of a vessel with an acceleration delay problem owing to insufficient torque from the main engine, it is necessary to consider alternative measures to avoid fatigue fractures of its shaft during the vessel life expectancy, which is 20 to 25 years.

(a)

(b)

Figure 5. (a) Vibration characteristics of existing viscous-spring damper. (b) Vibration characteristics of the optimum viscous-spring damper.

Figure 6. (a) Synthesized torsional stress of the intermediate shaft. (b) Synthesized torsional stress of the propeller shaft.

4.2. Modified Design of Viscous-Spring Damper Optimized for a Propulsion Shafting System with an rpm Acceleration Problem

In this research, an alternative measure to avoid fatigue fracture is considered: modification of the damper design to reduce torsional stress below the limit for continuous operation. First, the dynamic characteristics of the damper and its shaft system were evaluated by applying modified damper parameters such as the stiffness coefficient (which has a deviation range of $\pm 15.0\%$) to the corresponding damping coefficient ratio based on optimum damper design theory. Table 3 shows that the stiffness coefficient of the damper selected as the optimum design theory of the viscous-spring damper is

changed by 5% and the optimum damping coefficient ratio is interlocked with it. Here, Case IV shows the viscous-spring damper with the stiffness coefficient and the damping coefficient ratio obtained according to the optimum damper design theory. In other cases, the stiffness coefficient is increased or decreased by 5% based on Case IV.

Figure 7 shows the dynamic characteristics of the damper obtained by modifying the stiffness coefficient and the damping coefficient ratio linked to the damper selected as the optimum design theory of the viscous-spring damper. Figure 8a,b shows the result of the varied torsional stress of shafts. These figures show that when the stiffness coefficient of the damper is increased or decreased by 5%, the magnification is increased by about 15%. Moreover, the torsional stress acting on the shaft decreases when the stiffness coefficient of the damper is increased, but the torsional stress is increased when the stiffness coefficient is decreased. In the case of the intermediate shaft, the torsional stress acting on the shaft can be reduced to less than the fatigue limit line when the stiffness coefficient of the damper is increased by more than 10% compared to Case IV. In the case of the propeller shaft, the torsional stress acting on the shaft can be reduced to less than the fatigue limit line only if the stiffness coefficient of the damper is increased by more than 15% compared to Case IV. Therefore, it can be seen that the BSR may not be set within the operating speed range if the stiffness coefficient of the damper is increased by 15% more than the stiffness coefficient of the damper selected according to the optimum design theory of the viscous-spring damper.

Normally, the torsional vibration of the shaft system can be controlled by means of the optimum damper based on optimum damper design theory. However, in the case of a vessel with an acceleration delay problem, it is better to eliminate the barred speed range in consideration of fatigue stability. As a result of reviewing the torsional vibration characteristics of the corresponding shaft system by redesigning the existing damper in accordance with optimum damper design theory, it is insufficient to reduce the torsional stress in the intermediate shaft below the fatigue limit τ_1 for continuous operation. In this case, it is possible to reduce the torsional stress below the fatigue limit line to achieve the continuous operation recommended by the classification society by adjusting the design parameters of the damper such as stiffness coefficient and damping coefficient ratio, rather than using the design based on optimum damper design theory.

Figure 7. Vibration characteristics of the viscous-spring damper according to varied stiffness and damping coefficient ratio.

(a)

(b)

Figure 8. (a) Synthesized torsional stress of intermediate shaft according to varied stiffness and damping coefficient ratio. (b) Synthesized Table 3. A case study with varied stiffness and damping coefficient ratio based on optimum damper design.

Table 3. Case study with varied stiffness and damping coefficient ratio based on optimum damper design.

Classification	Case I	Case II	Case III	Case IV	Case V	Case VI	Case VII
Stiffness (Kd) MNm/rad	6.08	6.44	6.80	7.16	7.51	7.87	8.23
Damping coefficient ratio (ξ)	0.16	0.16	0.15	0.14	0.14	0.15	0.16

5. Conclusions

In this study, the vibration characteristics of a fuel efficient ultra-long-stroke engine using de-rating technology were reviewed and dynamic characteristics of a viscous-spring damper used to control the torsional vibration of its shaft system that had recently exhibited an acceleration delay problem was also investigated. These were examined in detail to determine what vibration characteristics might be used to optimize the viscous-spring damper to cope with the acceleration delay phenomenon. Then suggestions were made for proper measures to control the vibration effectively. The following conclusions were derived from this study.

- (1) Fuel efficient ultra-long-stroke engines have higher mean torque in the low-speed range to increase propulsion efficiency by adopting low speed and a large-diameter propeller. However, this results in an increase of the vibratory torque up to 40% as the stroke is increased by 16%. In addition, the engine de-rating technology used to improve fuel efficiency causes the mean effective pressure to decrease. This also results in an increase of the torsional exciting force at corresponding mean effective pressures.
- (2) The BSR is set with a prerequisite for rapid passage, and generally the engine passes this range within five seconds. However, if the de-rating ratio applied to improve fuel efficiency is excessive, the torque limit line is lowered parallel to that of a standard engine, causing a decrease in the engine acceleration performance in the low load range. Consequently, it can take more than one minute for the propulsion shafting system of the ship to pass through this range. In such cases, there is a possibility of fatigue fracture, such as that resulting from a reduction of the shaft fatigue life.

In general, the torsional vibration of the shaft system can be controlled effectively by optimizing the torsional vibration damper based on the optimum damper design theory. However, in the case of shaft systems with acceleration delay problems, it is better to eliminate the barred speed range in consideration of shaft fatigue stability. However, it is insufficient to reduce the torsional stress in the intermediate shaft below the fatigue limit τ_1 for continuous operation by using the viscous-spring damper according to the optimum damper design theory.

- (3) In order to eliminate the BSR of the propulsion shafting system of ships that exhibit an acceleration delay problem within the operating speed range, it is definitely desirable to adjust the design parameters of the torsional vibration damper considering the torsional vibration characteristics of the corresponding shaft system, instead of applying the design resulting from optimum damper design theory, to lower the torsional stress of its shaft below the fatigue limit to achieve the continuous operation recommended by the classification societies.

Author Contributions: Investigation, Resource, Writing-original draft, J.J.; Funding, Acquisition, Data curation, Writing-review & editing, C.K.; Data curation, Formal analysis, Investigation, Project administration, Software, Writing-original draft, Y.K. All authors have read and agreed to the published version of the manuscript.

Funding: This work was supported by a grant from the National R&D Project of “Development of LNG Bunkering Operation Technologies based on Operation System and Risk Assessment” funded by Ministry of Oceans and Fisheries, South Korea (PMS4310). All support is gratefully acknowledged.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

COT	Crude oil tanker
DLF	Dynamic limiter function
IMO	International maritime organization
MIP	Mean indicated pressure (bar)
MEP	Mean effective pressure (bar)
M.O.I	Moment of inertia (kgm^2)
SMCR	Specific maximum continuous rate (kW)
VLCC	Very large crude oil carrier
VLOC	Very large ore carrier
C	Damping coefficient
c_E	Damping coefficient of the main vibration system
c_d	Damping coefficient of damper
J	Mass moment of inertia
J_E	Moment of inertia for damper outer ring
J_d	Moment of inertia for damper outer ring
K	Stiffness coefficient
K_E	Torsional stiffness of main vibration system
K_d	Torsional stiffness of the damper
M_R	Dynamic magnitude at $\mu_E = 0$
M	Dynamic magnitude
Q	Exciting force

Greeks symbols

θ	Torsional angle
θ_E	Angular amplitude of the main vibration system
θ_{st}	Static angular displacement of the main vibration system
θ_d	Angular amplitude of damper outer ring
μ_E	Damping coefficient ratio of main vibration system
λ	Natural frequency ratio
η	Forced frequency ratio of the main vibration system
ω	Angular frequency
ω_E	Natural angular frequency of main vibration system
μ	Optimum damping coefficient ratio
ξ	Damping coefficient ratio of viscous-spring damper
τ_1	Fatigue limit value of shaft (N/mm^2)
τ_2	Limit value of yield strength of shaft (N/mm^2)

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